Experimental identification of noise reduction properties of honeycomb panels using a small cabin

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Abstract

A procedure to identify the noise reduction properties of panels by means of a single cabin test setup is discussed. The complexity of the sound pressure field that builds up in the acoustic volume requires the support of advanced numerical techniques allowing for the evaluation of noise and vibration performances. Numerical models are used to predict the structural dynamics and the vibro-acoustic behaviour of the tested panel. Both Finite Element and Boundary Element simulations are validated by means of the experimental tests. The proposed test methodology is mainly devoted to the case of medium and small sized lightweight panels, although generally applicable to a much wider set. An aluminium honeycomb core sandwich panel for aerospace application is investigated in the present work. Modal analyses and sound Insertion Loss tests are used to verify the structural model, which is afterwards implemented to predict the standard sound Transmission Loss of the analysed sample.

Keywords: noise reduction; lightweight structures; vibro-acoustics; experimental test rig.

Résumé

Une procédure pour identifier les propriétes de réduction sonore d'un panneau via un dispositif du type 'petite cabine' est présentée. La complexité du champ de pression dans le volume acoustique nécessite le support de techniques numériques avancées pour permettre l'évaluation des performances bruit et vibrations. Des modèles numériques sont utilisés afin de prédire le comportement dynamique et vibro-acoustic du panneau étudié. Des simulations éléments finis et éléments frontières sont validées au moyen des mesures. La méthodologie expérimentale proposée est principalement adaptée dans le cas de panneaux légers de petites et moyennes tailles, bien que généralisable à un plus grand nombre. Un paneau-sandwich en aluminium à cœur alvéolaire à application aérospatiale est étudié dans le présent document. Des analyses modales expérimentales et des mesures d'Insertion Loss sont utilisées pour vérifier le modèle structurel, lequel est implémenté après afin de prédire le Transmission Loss standard de l'échantillon étudié.

Mots clés: réduction de brut, structures légères, vibro-acoustique, dispositif expérimental

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1. Introduction

The reduction of CO_2 emissions has become worldwide a critical issue. A significant effort is specially made on the enhancement of means of transport, mainly automotive railway and aerospace, taking into account that this industrial sector contributes for more than 25 % to the entire CO_2 production (M. Goede, M. Stehlin, L. Rafflenbeul, G. Kopp, E. Beeh, 2008). Lightweight solutions find here broad application, offering an inherent high stiffness-to-mass ratio. This can in fact directly translate into substantial benefits in terms of weight and energy savings, still offering high mechanical performances. The enhanced performance and efficiency can clearly make the final product more competitive. On the other hand, the high stiffness-to-mass ratio of lightweight components may result in unsatisfactory vibro-acoustic behaviour, impairing the Noise, Vibration and Harshness (NVH) performances. Sometimes additional post-treatments are adopted, consisting of the addition of layers of visco-elastic materials, which can cause a significant reduction of the initially expected weight gain. Hence, good a-priori knowledge of NVH properties of the applied solutions is of crucial importance. Both experimental and numerical tools can efficiently be implemented at this purpose.

Amongst others, the sound Transmission Loss index (TL) is used to indicate the acoustic attenuation properties of a given structure. The sound transmission loss measurement provides an experimental characterization of the sound transmission performance of a test partition. The TL measurements require dedicated test facilities, the design of whom is indicated by international standards (ASTM Standard E90 04 in 2009). Normally very big volumes are involved. The same specimen tested in different facilities will normally show the same behaviour (as far as each of them fulfils the requirements prescribed in the aforementioned standard).

Typically the test rig consists of two adjacent rooms:

- A reverberant source room (for random incidence airborne excitation)
- An anechoic receiving room (to simulate free field radiation)

The test sample - of a minimum dimension of 1.2 m (except for the thickness) - is mounted at the rigid partition wall between the two rooms. A "diffuse" sound field is generated in the source room. The uniform sound pressure level and the equally probable flow of energy in all directions cause the test specimen to vibrate. This creates a sound field in the second room. The Sound Pressure Levels (SPL) in both the source and receiving rooms are recorded (ASTM Standard E90 04 in 2009) and respectively called $P_{incident}$ and $P_{transmitted}$. The TL comes from Eq.1.

$$TL = 10Log_{10}\frac{P_{incident}}{P_{trasmitted}} \tag{1}$$

Care has to be taken when measuring the pressure field in the reverberant source room, avoiding direct field effects and cavity resonances (satisfying the hypotheses of field diffusivity). In the case a reverberant receiving room is also used, the same care is also required for the corresponding field acquisitions and Eq.1 needs to be corrected according to the Sabine's formula (M.Vivolo, 2013). Relying on the assumptions of diffuse field and/or free field, the method is straightforward in terms of the parameter identification procedures. On the other hand however, the whole facility can be quite demanding and cumbersome, and the required sample dimensions can sometimes result impractical especially for small and medium sized parts.

A new experimental test rig was designed at KU Leuven, with the aim to get a faster and more practical tool to test and evaluate noise reduction properties of lightweight panels of common use in the small sized transport industry (M. Vivolo, B. Pluymers, D. Vandepitte, W. Desmet, 2011). The test rig, named PMA Soundbox, supports the research and development of new materials and the optimization of lightweight component designs. Unlike standard facilities, the PMA Soundbox is a multifunctional test setup, which can be used for measuring several NVH parameters, i.e. sound transmission of panels, sound absorption of porous materials, vibration damping and it can also support material properties identification. Compared to the case of standard sound TL measurements, the experimental test procedure does not rely on simplified field assumptions, but a more complex pressure field is present. Hence, in the low frequency region (below field diffusivity), the experimental identifications are supported by numerical modelling techniques able to tackle coupled vibro-acoustic problems, characterized by modally dominated acoustic sub-domains, which are typically encountered in transport industry.

In this work the PMA Soundbox is exploited to investigate the noise reduction properties of an Aluminium honeycomb core sandwich panel used in transport vehicle design. The present paper is structured in three main sections. First, a general overview is given about the design and technical specifications of the setup. The second part reports about the tests carried out for vibro-acoustic studies of the tested panel. These experimental results are compared with numerical simulations. Modal analysis data are used to validate the Finite Element (FE) model of the sandwich plate. This structural numerical model is coupled to a Boundary Element (BE) model of the acoustic cabin, in a fully coupled problem, and the numerical results compared to the experimental measurements. In the last section, the noise reduction performance measured with the PMA Soundbox is compared to the TL evaluated by means of standard facility. Ongoing and upcoming studies, aiming at widening the implementation of the PMA Soundbox in the research field of the vibro-acoustic study of lightweight structures, are finally given in the last section.

2. The vibro-acoustic test rig

The design of the PMA Soundbox test rig is based on a preliminary work carried out on a smaller setup (M. Vivolo, B. Pluymers, D. Vandepitte, W. Desmet, 2010) and consists of a single cabin. The setup offers the advantages of being relatively small (only 0.83 m^3), moveable and cheap. It allows for several kinds of measurements on components of different sizes and thicknesses in a wide frequency range ([50-20000] Hz), for both airborne and structure-borne excitation. The test window can be enclosed by a flexible component. In this configuration, the sound Insertion Loss (IL) can be measured. The sound IL is defined as the difference in dB between the SPL radiated outside the cabin when an acoustic source is active inside, in the so called open-window configuration, P_{open}, and the SPL radiated when the tested component is mounted, P_{closed}.

$$IL = 10Log_{10}\frac{P_{open}}{P_{closed}} \tag{2}$$

Unlike the TL, the IL not only depends on the NVH properties of the examined structure but it is also a function of the coupled system, consisting of both the vibrating element and the acoustic cavity. Although the test procedure is not very demanding, care is required in the discussion of the sound transmission characteristics of the test panel, especially in the "low" frequency region. Below field diffusivity, the modal behaviour of the acoustic subdomain strongly couples to that of the structural component. In view of this, the PMA Soundbox has been designed in order to get a uniform distribution of the natural acoustic frequencies, avoiding overlapping frequencies and uneven gaps between subsequent acoustic natural frequencies. To this purpose, the final design of the test cabin consists of a non-rectangular parallelepiped volume, in which the inner walls are not parallel. The global shape is convex (C. Gonzales Diaz, M. Vivolo, B. Pluymers, D. Vandepitte, W. Desmet, 2010). The simplicity of the geometry and the accuracy of the imposed boundary conditions allow for accurate numerical models, able to satisfactory predict the observed behaviour over a wide frequency range.

Fig. 1 shows the practical realisation of the PMA Soundbox as a five walled reinforced concrete cabin. The cabin can be enclosed at the front side by a modular partition. Test panels of different thickness and in-plane dimensions can be installed using different front partition walls. All the available front walls are made of aluminium and each of them is 0.035 m thick (see Fig. 1 (a)).

The available test partitions are listed below, sorted by the dimension of the test windows:

- A1 window
- A2 window
- 500 mm x 350 mm window
- A3 window
- A4 window
- Fully closed wall

The total weight of the setup is about 3 tons. A three wheels system allows the rig to be moved to the desired location. As visible in Fig. 1 (a), the PMA Soundbox is suspended on a four air-spring system, required to suppress vibrations coming from the laboratory floor.

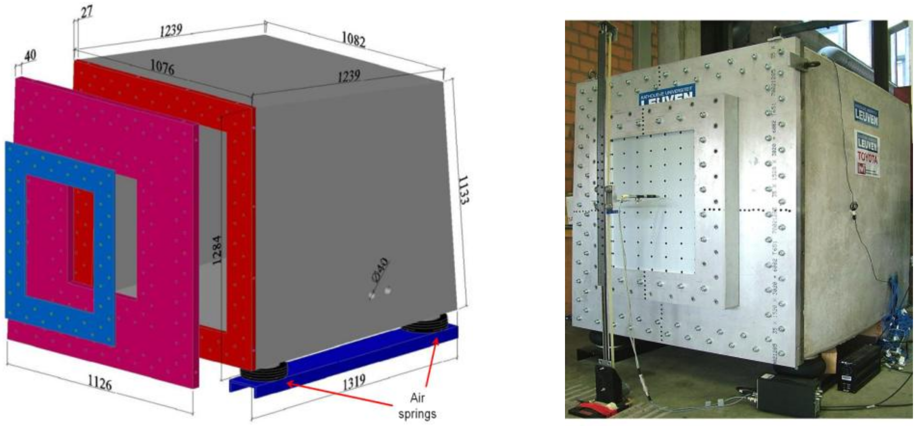

Fig.1. (a) A sketch of the vibro-acoustic test rig; (b) Sound intensity measurements

The setup is equipped with several sensors for acoustical and structural response acquisitions (a PP sound intensity probe, free-field and reverberant-field ½" microphones, lightweight 1D accelerometers). By means of a full range speaker set inside the cabin or of an electro-dynamic shaker or an impact hammer, both structure-borne and airborne excitations can be provided.

3. Sound transmission characterization of an aluminium honeycomb core sandwich panel

In this section, an aluminium honeycomb core sandwich panel used for space rocket applications is presented. In collaboration with the Japanese Aerospace Agency (JAXA), the noise reduction performances have been experimentally and numerically tested, following both the standard test methodology and by using the PMA Soundbox.

3.1. Honeycomb sandwich tested panel

The tested specimen is a honeycomb sandwich plate of 11.2 mm total thickness. The core is an AL-3/16-5052T-.001, 10 mm thick and made out of hexagonal regular cells. The top and bottom face sheets are 0.6 mm thick and made out of AL-5052T. The geometry is schematically represented in Fig.2 and the main geometric dimensions are listed in Table 1.

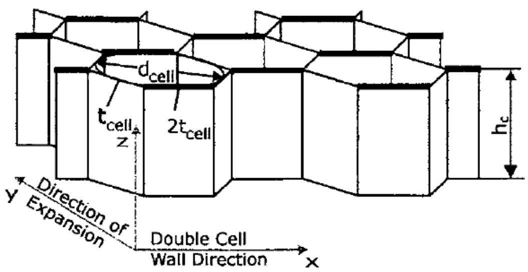

Fig.2. Geometric configuration of the core of the tested honeycomb sandwich panel

3.2. Experimental tests

Modal analyses in free and clamped edges conditions have been performed, in order to verify the numerical model of the tested panel. Then, sound intensity tests have been carried out to investigate its noise reduction properties.

Table 1. Geometric properties of the tested honeycomb sandwich panel

Geometric properties of the tested panel	mm
t_{cell}: wall thickness	0.034
d_{cell}: cell size	4.76
h:core height	10
t_s: surface sheets global thickness	1.2
length along direction of expansion in free edges condition	600
length along double cell wall direction in free edges condition	800

3.2.1. Modal analyses

Rowing hammer impact tests have been performed in free and clamped edges conditions. A test grid has been defined in order to capture the modal shapes up to a maximum of half wavelengths equal to 4 along the major edge (chosen as y axis) and 3 along the shorter one (chosen as x axis). The grid is made of 88 input points and three of them have been also used as output points. Three lightweight accelerometers (pcb 352A24) have been placed to record the acceleration signals at these locations. In free edges condition the panel is suspended using a spring and its size is 600 x 800 mm^2. In Fig.3 the sum of the three measured FRFs shows the natural frequencies up to 1 kHz and the first five experimental modal shapes are indicated.

When the sandwich panel is tested with clamped edge conditions, the test surface is reduced to an A2 size, e.g. 420 x 594 mm^2. In order to better identify the uncoupled structural natural frequencies of the tested panel, the experimental modal analyses in clamped condition have been performed in two different configurations:

- The isolated system, consisting only of the front wall and the panel clamped at the test window
- The coupled system, having the tested panel clamped to the front wall and mounted on the PMA Soundbox

The sums of the FRFs recorded in the two cases are compared in Fig. 4 where the first clamped modes are also shown. Considering the isolated system with the front wall resting vertically on the floor and the honeycomb panel clamped at the test window, the sum of the FRFs shows the natural frequencies of the clamped panel. Some extra peaks related to the modal behaviour of the front wall itself can be also observed. The amplitude of these occurrences is however not comparable to that of the natural modes of the panel itself.

When the front wall is mounted on the Soundbox, the influence of the acoustic subdomain is evident. The acoustic and the structural natural modes couple together and give rise to a multitude of peaks that are visible in the sum of the FRFs. A shift towards higher frequencies of the natural coupled frequencies can be observed compared to the isolated system. The above mentioned shift can be related to the higher stiffness that the system reaches when the front wall is coupled to the cabin.

3.2.2. Insertion Loss measurements

The sound IL of the honeycomb panel is measured with the PMA Soundbox. This requires two measurement sets:

- The measurement of the acoustic power radiated outside the cabin when the test window is left empty (open-window configuration)
- The same measurement repeated with the honeycomb panel mounted at the front wall (closed-window)

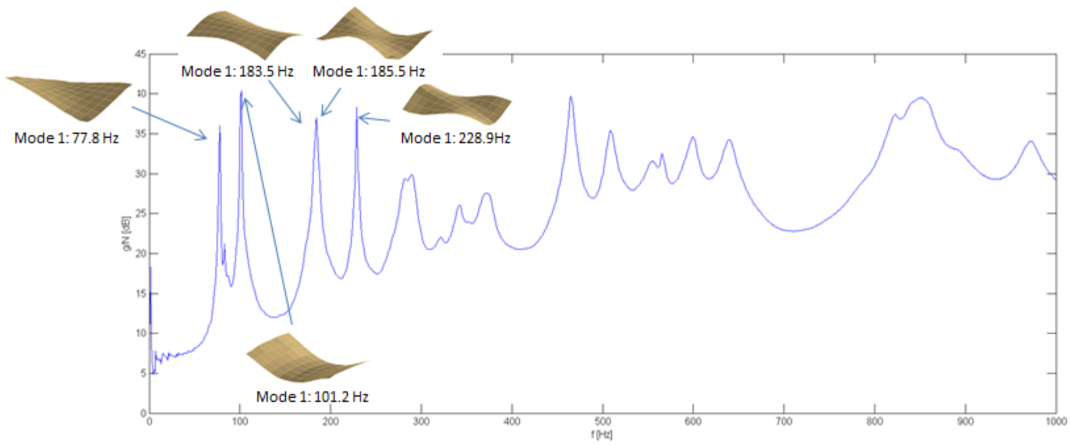

Fig.3. Experimental sum of FRFs function recorded on the tested panel (88 points) in free conditions and first five modal shapes

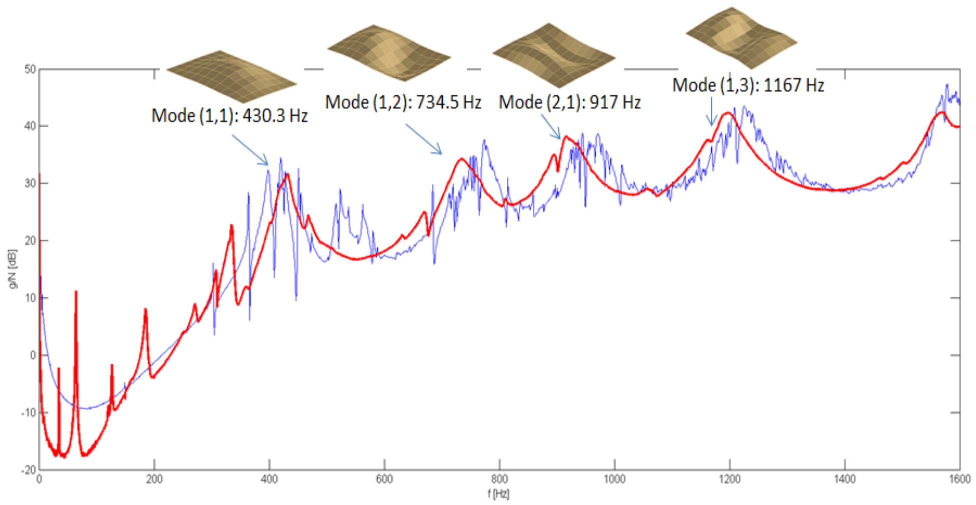

Fig.4. Experimental sum of the FRFs function recorded on the tested panel (88 points) clamped to the Soundbox (blue curve) and clamped only to the frame (red curve)

In both cases, the inner pressure field is excited by a broad range loudspeaker fed by a white noise voltage signal up to 1600 Hz. The speaker is placed at one of the inner corners of the cabin in order to excite all the acoustic modes. The acoustic power radiated outside is obtained from the sound intensity measured over the test area by means of a PP sound intensity probe. The probe scans an area 10 cm far from the tested panel. The sound intensity is measured in 70 points. Each of them corresponds to the centre point of an element of the grid previously used for the modal hammer test with clamped edges condition. The sound intensity detected by the PP probe at each output point is considered to be the averaged value over the corresponding elementary surface (Fig.1b).

The third octave band averaged sound IL is plotted in Fig.5 together with the sound intensity mapping over the acquisition plane in front of the test area at the frequencies of the first modes. Up to 1 kHz the strong modal low frequency behaviour is clearly indicated by the presence of well separated dips in the IL

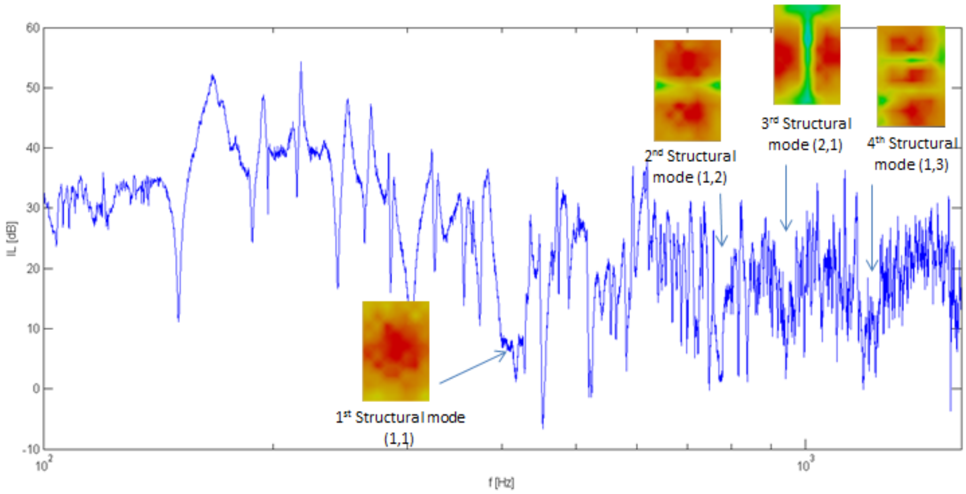

Fig.5. Experimental sound IL of the honeycomb sandwich panel. A2 size, clamped edges

3.3. Numerical analyses

A 2D multi-layered FE model of the honeycomb sandwich panel is built using MSC.Patran 2012. The size of the edges of each plate element of the model is 5 mm in order to have a convergence of the modal analysis results up to 3200 Hz. Two plies are used to represent the homogeneous Aluminium skins, while a layer with 2D orthotropic elastic properties is placed between them simulating the honeycomb core (Gibson & Ashby, 1999). The numerical modal analyses are computed using MSC Nastran 2011.1, while the experimental-numerical correlation is evaluated using Virtual.Lab Rev11.

3.3.1. Numerical free edges modal analysis

The experimental-numerical correlation between the first ten modes of the freely suspended modal analyses is reported in terms of the Modal Assurance Criterion (MAC) in Table 2. The MAC provides a measure of consistency (degree of linearity) between estimates of a modal vector. It is a quality assurance indicator for experimental modal vectors that are estimated from measured frequency response functions. The high values of the same order modal pairs and the small frequency difference (on average less than 5%) indicate a good correlation between the experimental test and the numerical model. The same model is used to check the correlation of the clamped panel case.

Table 2. Experimental-numerical modes correlation table for freely suspended modal analyses - 600mm X 800 mm honeycomb sandwich panel

Exp. Mode ID	Exp. Frequency [Hz]	Num. Mode ID	Num. Frequency [Hz]	MAC Value	Num.Freq.-Exp.Freq. (% of Exp.Freq.)
1	77.8	1	85.5	0.988	9.9
2	101.2	2	103.3	0.986	2.1
3	183.5	3	183	0.727	-0.3
4	185.5	4	198.6	0.935	7
5	228.9	5	237.6	0.974	3.8
6	284.8	6	297	0.906	4.3
7	365.2	7	372.8	0.981	2.1
8	373.7	8	390	0.978	4.4
9	465.2	9	473.4	0.984	1.8
10	509.4	10	528.8	0.974	3.8

3.3.2. Numerical clamped edges modal analysis

Similar to the free conditions case, the structural modal behaviour is well predicted by the multi-layered 2D model for the clamped conditions. Table 3 shows the good agreement between numerical and experimental mode shapes and natural frequencies for the clamped configuration.

Table 3. Experimental-numerical modes correlation table for clamped edges condition modal analyses – A2 clamped honeycomb sandwich panel

Exp. Mode ID	Exp. Frequency [Hz]	Num. Mode ID	Num. Frequency [Hz]	MAC Value	Num.Freq.-Exp.Freq. (% of Exp.Freq.)
1	430.3	1	484	0.946	12.5
2	734.5	2	763.3	0.957	3.9
3	917	3	1000.5	0.941	9.1
4	1167	4	1187.4	0.871	1.7
5	1227.6	5	1228.1	0.945	0.04

3.3.3. Numerical IL analysis

In order to numerically investigate the IL of the tested panel, a BE model of the cabin and the clamped honeycomb panel is used. Finite acoustic impedance is imposed at the inner walls of the cabin. The frequency independent real value of 140k kg/m^{2}s, corresponding to the acoustic impedance of typical polished concrete (Norton, M., and Karczub, 2003), is considered in order to take into account the finite (although small) sound absorption ability of these surfaces. Dissipative phenomena are also considered in the complex speed of sound, which represents the energy loss of the sound waves during their travel, more effective especially at higher frequencies. Structural damping is discarded.

Fig.6 shows the comparison in narrow bands between experimental and numerical IL. A good prediction can be observed in the whole frequency range, from 150 Hz up to 1600 Hz, except in the band from 400 Hz up to 500 Hz. In this region, the first structural mode of the clamped panel occurs. The non-perfect clamped boundary conditions realized in practice have a big influence especially on the lower order modes. The first experimental mode occurs at 430 Hz, while numerically predicted at 480 Hz. Below 150 Hz the characteristics of the loudspeaker are too poor to provide a flat input spectrum. The discrepancy between 200 Hz and 250 Hz is due to the non-perfect clamping conditions of the real panel, whereas in the numerical model the panel is fully clamped. This in combination with a predominantly stiffness dominated panel response at these frequencies results in a mismatch of the predicted and measured IL.

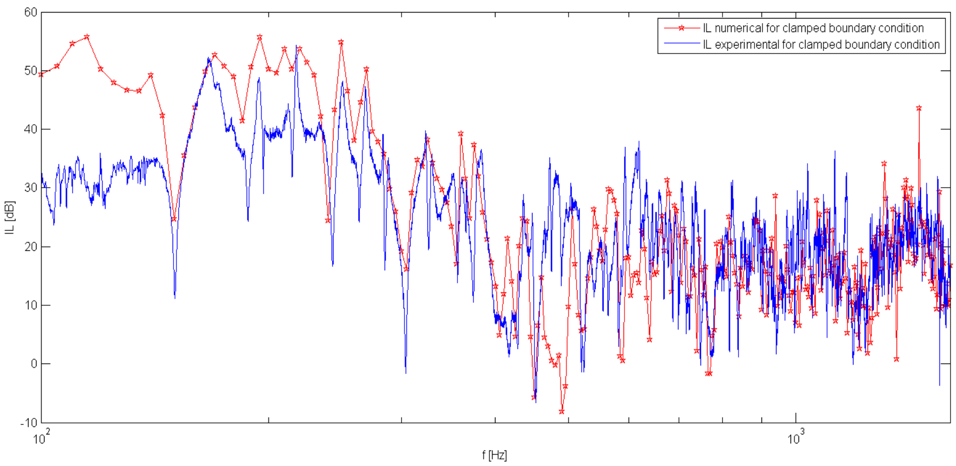

Fig.6. Experimental and numerical IL (narrow frequency) – A2 clamped honeycomb sandwich panel

3.4. TL evaluation

During the sound IL test, the Sound Pressure Level (SPL) inside the Soundbox can also be recorded by means of microphones placed inside the cavity. Their positions are chosen in order to avoid the influence of the interference phenomena close to the cavity's walls (Fig.7).

Fig.7. Interior view of the Soundbox

Considering the closed-cabin configuration, the difference between the internal averaged SPL and the externally radiated SPL is named here normalized radiated power. Fig.8 shows the comparison between the experimental normalized acoustic power and the experimental TL, measured by Japanese Aerospace Agency (JAXA), of the same honeycomb sandwich panel but with a different size of 700*1000 mm^2.

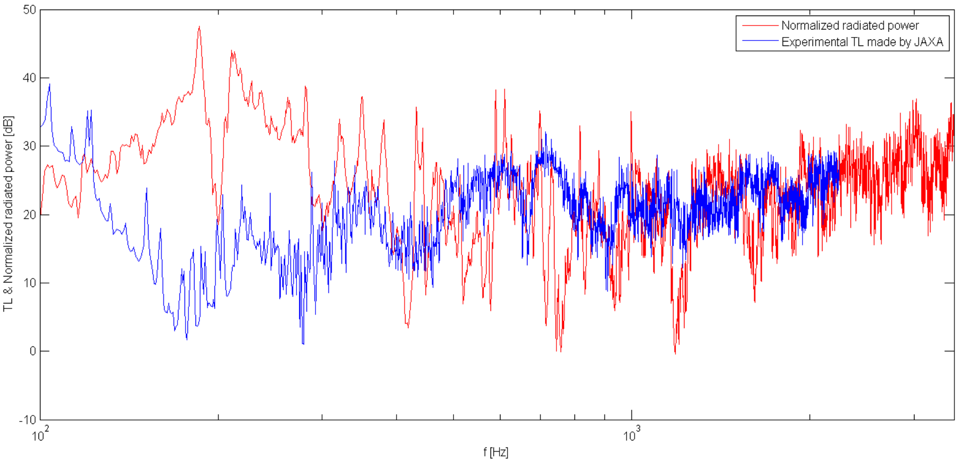

Fig.8. Normalized radiated power measured at KU Leuven – A2 panel (red curve); Standard TL measured by Japanese Aerospace Agency (JAXA) – 700 mm X 1000 mm panel (blue curve)

The Soundbox reaches a diffuse field above 1600 Hz and, as shown in Fig.8, here there is a good matching between the two curves, while below this frequency diffusivity is lost inside the Soundbox and the acoustic modal loading is becoming apparent, illustrating the value of the new test rig for vibro-acoustic characterization of panels with application in similar environments.

Combining the above results with the correlations results from section 3.3.3. shows the high potential of the test rig to become a valuable tool for vibro-acoustic characterisation of lightweight panels under structural/acoustic loading and to determine standard TL values by means of non-standard measurements in a non-standard environment.

4. Conclusions

A new experimental setup to study the vibro-acoustic behavior of lightweight structures is presented and discussed. Structural-acoustic coupled characteristic, coincidence frequency and acoustic IL can be measured in a fast and easy manner. The testing apparatus is designed to be small, movable, and consequently also able to provide repeatable measures and a desired and predictable low frequency acoustic field. The capability to have predictable acoustic characteristics of the presented facility, also in the fully modally dominated frequency region, become appealing for investigating noise and vibration properties of lightweight components in a broadband frequency range. The aim is to extract useful parameters from the measured data, coming from a non-standard tool, in order to get a reliable prediction of the standard noise and vibration properties of lightweight panels themselves. Furthermore, the simplicity of the geometry does not require very sophisticated numerical models to provide satisfactory predictions of the measured behaviour in a very wide frequency domain. A wave based model can be easily prepared to this end to overcome the limitations of standard numerical techniques (FEM\BEM) at higher frequencies. This setup has been used to test a lightweight component, such as a honeycomb sandwich plate, in order to investigate its noise and vibration characteristics. Future works will focus on health monitoring and structural damping detection of lightweight structures.

5. Acknowledgements

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